AMENDMENTS TO THE DRAWINGS

Attached are three (3) replacement sheets of drawings for Figures 1, 2, 3A and 3B as originally filed. Also attached are four (4) New Sheets which contain Figures 4A, 4B, 4C and 5.

Attachment: Replacement Sheets (3)
New Sheets (4)

REMARKS/ARGUMENTS

Claims 1-19 are pending. By this Amendment, claims 9, 10 and 14-16 are amended, and new claims 17-20 are added.

In addition, a Substitute Specification is provided herewith which includes amendments to address the various objections and rejections set forth in the Office Action. No new matter is included in the Substitute Specification. Furthermore, the original drawings are replaced with new Figures 1-3B provided herewith. Further, new Figures 4A-5 are provided herewith.

Reconsideration in view of the above amendments and the following remarks is respectfully requested.

In paragraph 1 of the Office Action, the incorporation of material by reference is objected to. By this Amendment, various drawings and materials are included herein to overcome this rejection. The specifics of what was added to the specification and the drawings is described more fully below.

Reconsideration and withdrawal of the objection are respectfully requested.

In paragraph 2 of the Office Action, the disclosure is objected to based on a minor informality on page 2, line 22 which has been corrected in the Substitute Specification. Withdrawal of the rejection is respectfully requested.

In paragraph 3 of the Office Action, the drawings were objected to since it is said that throughout the specification Applicant makes reference to drawings in a foreign application and such drawings must accompany this application. By this Amendment, Figures 4A, 4B, 4C and 5 are added. Figures 4A-4C are reproductions from Figures 1-3 of WO 00/46486 and, in conjunction with the amendment to the specification provided herewith, help better explain the claimed eccentric oscillation of the drive shaft, as recited in claim 1. In addition, new Figure 5 is

provided herewith which is a copy of Figure 7 of WO 00/46486. Various aspects of this drawing are referred to in the Background of the Invention.

Reconsideration and withdrawal of the objection to the drawings are respectfully requested.

Claims 1-16 were rejected under 35 U.S.C. §112, first paragraph. This rejection is respectfully traversed as Figures 1-3 from WO 00/46486 are incorporated herein as new Figures 4A-4C provided herewith. It is noted that WO 00/46486 was incorporated by reference in the Background Section of the Invention. In addition, the corresponding description of these figures has been taken from the text of WO 00/46486 and appears in the attached Substitute Specification. Again, no new matter has been entered into the Substitute Specification. Only matter which was originally disclosed in WO 00/46486 has been added.

Reconsideration and withdrawal of the rejection are respectfully requested.

Claims 8-16 were rejected under 35 U.S.C. §112, second paragraph. By this Amendment, the claims are amended for clarity only. For example, claim 9 specifies that the second bearing is a pressurized fluid lubricated bearing.

In claim 11, the term "in" has been eliminated to obviate that objection.

Furthermore, claims 14-16 have been amended to specify the limited rotational speed of the cutting disc, when free running, is within a certain RMP range.

It is noted that claim 8 is rejected since it allegedly does not have antecedent basis for "the fluid bearing". However, claim 8 does not recite a fluid bearing, but instead recites a hydrostatic bearing, for which the antecedent basis is provided in claims 7 and 5.

Reconsideration and withdrawal of the rejection are respectfully requested.

Claim 1 was rejected under 35 U.S.C. §102(b) over Snyder (U.S. Patent No. 5,125,719). This rejection is respectfully traversed. At the outset, the Office Action provides an extremely scant description of Snyder, only stating that "Snyder discloses an oscillating disc cutter comprising a radial bearing (50, 52), first axial bearing (80)." Therefore, the Office Action fails to address much of the language in claim 1, e.g., that the drive mechanism includes a drive shaft to effect eccentric oscillation of the cutting disc and a radial bearing disposed to permit relative rotation between the drive shaft and the cutting disc. The Office Action also does not address the language in claim 1 that specifies that the cutter further includes a first axial bearing disposed to react axial forces while accommodating induced rotation of the cutting disc when operatively engaged and to induce a rotational drag thereby limiting rotational speed of the cutting disc when free running.

Moreover, Snyder does not teach an oscillating disc cutter as recited in the preamble of claim 1. Snyder is directed toward a tunnel boring machine (TBM) having increased thrust capability by means of a hydrostatic bearing located between the cutting head and the thrust cylinders which transmit forward axial thrust to the cutter head. TBMs have a large circular cutting head mounted on a drive shaft 38 disposed for rotation about a horizontal axis. The head includes a plurality of cutting discs 16 for engaging the rock face. The cutters are of the rolling type, disposed on the head to roll around a tangential cutting path under enormous axial pressure. The higher the thrust capacity of the machine, the faster it can cut. However, because the head must rotate to cut, pressure must be exerted through bearings, which are subjected to the tremendous axial thrust loads. Snyder states that the load bearing capability of these axial thrust bearings is the common limitation of TBMs cutting capability. See column 1, lines 10-34. As a solution, Snyder proposes replacing the conventional mechanical type axial bearings in the TBM

with a pressurized fluid bearing. The ability of fluid-type bearings to support far greater axial load than conventional bearings is well known.

It should be particularly noted that the head is connected to the drive shaft to permit torque transmitting drive. Rotation of the shaft and head is achieved by a mechanical gear linkage to the motor. See column 3, lines 64 – column 14, line 1. As such, the speed of the head is directly linked to the motor and over speeding of the head is not an issue.

As such, Snyder does not teach or suggest the subject matter of claim. First, Snyder does not disclose an oscillating disc cutter as defined in claim 1. Instead, Snyder teaches a TBM which relies on rolling cutters to effect cutting. Applicants believe that the Examiner should be aware of the differences between TBMs (rolling cutters) and oscillating disc cutters as presently defined in claim 1, especially since Examiner Singh is also handling Applicant's U.S. Application No. 09/889,745 for a "Rock Boring Device".

In addition, Snyder does not disclose a drive mechanism including a drive shaft to effect eccentric oscillation of the cutting disc as defined in claim 1. Snyder includes a drive shaft but there is no mention that it is used to provide oscillatory motion.

While Snyder may include discussion of radial bearings to support the shaft, there is no mention that these radial bearings permit relative rotation between the drive shaft and the cutting disc. Rather, the radial bearings allow for rotation of the shaft, which is connected by a direct torque-transmitting link to the head. In contrast, the radial bearing defined in claim 1 allows the cutting disc of the oscillating disc cutter to be free to rotate on the shaft. Snyder may also include discussion of an axial thrust bearing to react to axial forces. However, there is no mention in Snyder that the axial disc be used to accommodate induced rotation when operatively engaged and to induce a rotational drag thereby limiting rotational speed of the cutting disc when

free running, as recited in claim 1. The concept of using a bearing in Snyder as a source of drag, or to limit rotation makes absolutely no sense whatsoever. Opposing the rotation of the cutting head would only consume additional power. Moreover, over speeding is not an issue in TBMs because the cutting head is controlled directly by the motor. In any event, the rotational speed is low in comparison to the shaft as disclosed in the present application, which can range up to 3,000 RPM.

Reconsideration and withdrawal of the rejection are respectfully requested.

Claims 2-13 were rejected under 35 U.S.C. §103(a) over Snyder in view of WO 00/46486 (WO '486). This rejection is respectfully traversed.

WO '486 is disclosed in the Background of the present application and is owned by the Assignee of the present application. The prior application discusses the over speeding problem associated with oscillating disc cutters (ODCs). For example, WO '486 clearly describes the cause of over speeding and the problem it creates. See page 10, line 17.

To summarize, an ODC relies on an oscillating motion of the cutting disc to cut rock. This motion is produced by eccentrically mounting the cutter disc to a rotating drive shaft. The speed of the shaft preferably is relatively high, e.g., around 3,000 RPM for the tool to be most effective. Of course, it is the oscillatory movement of the cutter that allows it to cut rock not abrasion from a rotation. The disc is not designed to engage the rock face spinning at high speed, so, in order to isolate the cutter from the rotation of the shaft, it is mounted to the shaft on a bearing and is free to rotate. This is bearing 609 in Prior Art Figure 5 attached hereto. However, since most bearings provide some bearing drag, if the cutter is not physically engaged with the rock face, eventually frictional forces (bearing drag) from the spinning drive shaft will cause the disc cutter to rotate at or about the same speed as the shaft. This is over speeding. In

those circumstances, when the cutting edge of the rotating cutting disc is then engaged with the stationary rock face, it experiences a substantial drag load tending to slow the rotation of the disc. In practice, the cutting disc can be slowed, almost instantaneously, from about 3,000 RPM, with significant wear or damage resulting to the cutting edge.

WO '486 proposes a solution whereby the disc is mechanically linked to the body of the disc cutter 601 by means of a gear arrangement 616. By providing such a mechanical link the speed of the disc is controlled so that wear or damage can be largely reduced or eliminated.

In view of the vastly different modes of operation between Snyder and WO '486, Applicant respectfully submits that one of ordinary skill in the art would not have been motivated to combine them.

As discussed above, one aspect of WO '486 is that it seeks to overcome a problem with over speeding. However, in contrast to WO '486, claim 1 defines the use of a fluid bearing, between the cutter and the stationary housing 603 to not only produce axial support for the disc and accommodate induced rotation, but also to limit the speed of the disc when free running by inducing a rotational drag. This unconventional use of a fluid bearing is nowhere described in WO '486 or Snyder. In WO '486, the axial bearings 605 and 606 are only used to provide axial thrust. Instead, as seen in Figure 5 attached hereto, the gear arrangement 616 operates to mechanically control the speed of the disc, particularly when free running.

The teachings of Snyder do not make up for the deficiencies noted above with respect to WO '486.

Reconsideration and withdrawal of the rejection are respectfully requested.

Claims 14-16 were rejected under 35 U.S.C. §103(a) over Snyder in view of Chandrasekaran et al. (U.S. Patent No. 5,462,364). This rejection is respectfully traversed.

In paragraph 12 of the Office Action, the Examiner has asserted that Chandrasekaran et al. teaches controlling rotational speed to drag force. The Examiner has indicated that this is disclosed in the Abstract of the cited document. Applicant respectfully disagrees and asks that the Examiner consider the whole document rather than basing an assessment of the document's relevance based solely on the Abstract.

The Examiner's attention is directed to the discussion of Prior Art when Chandrasekaran et al. discloses a problem with existing hydrostatic bearings. In particular, at column 1, lines 23-27, Chandrasekaran et al. states that existing hydrostatic bearings cannot be used under heavy industrial loading at varying rotational spindle speeds where the range between the maximum and minimum intended use speed is greater than 10,000 RPM. Moreover, that hydrodynamic bearings inherently are unstable at low speeds due to insufficient dynamic pressure if the spindle loading is heavy; at high spindle speeds, i.e., 20,000-40,000 RPM, the fluid bearing may become thin due to heat and effective bearing performance. See column1, lines 38-42.

These acknowledged problems become the object of the patent disclosed in the last paragraph of the discussion of prior art beginning at column 1, line 53 which states: "what is needed is an integrated fluid bearing system than can attain at least 1,000 PSI hydro-dynamically at higher speeds (i.e., 10,000-40,000 RPM) by retaining closed hydrostatic pressures at lower speeds, thereby providing enhanced stiffness for a spindle subject to large side or thrust loads even at a variety of speeds."

Clearly, this document is directed to providing hydrodynamic bearing operable at a large speed range. It should be noted that this range is well outside the range disclosed in the present application, and claims 14-16 in particular.

Reading the specification shows that Chandrasekaran et al. proposes a hydrodynamic bearing designed to operate in one of two separable hydrodynamic modes depending on the speed of rotation. This is achieved by providing a bearing surface interrupted by a plurality of circumferential spaced pressure generating zones, such that at higher speeds, each zone is capable of dynamically compressing a fluid body contained in the zone into a crevasse in either direction of rotation of the spindle. At lower speeds, pressurized fluid is supplied to flow through each zone and out of the crevasse while exerting a predetermined hydrostatic force against the spindle.

This design is related to maintaining an operable pressure in the bearing at low speeds whilst preventing fluid thinning at high speeds. No where in this document, nor in the Abstract thereto, is any discussion of using the hydrodynamic zones to provide a drag in the bearing to limit the speed of the spindle.

Accordingly, Chandrasekaran et al. does not teach or disclose the RPM ranges recited in claims 14-16. Chandrasekaran et al. does not provide any discussion of using friction in a radial bearing to control rotational speed. Rather, Chandrasekaran et al. is concerned with accommodating a wide range of shaft rotational speeds and maintaining an adequate bearing stiffness. Again, as in Snyder, it makes no sense to limit the rotational speed of the shaft with the bearing because the shaft is directly powered to rotate.

Reconsideration and withdrawal of the rejection are respectfully requested.

In view of the above amendments and remarks, Applicant respectfully submits that all the claims are patentable and that the entire application is in condition for allowance.

Should the Examiner believe that anything further is desirable to place the application in better condition for allowance, the Examiner is invited to contact the undersigned at the telephone number listed below.

Respectfully submitted,

NIXON & VANDERHYE P.C.

By: ______________________
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PTB:jck
Attachments:
Marked-up Specification
Substitute Specification
Replacement Figures 1-3
New Figures 4A-5

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TITLE: OSCILLATING DISC CUTTER WITH SPEED CONTROLLING BEARINGS

FIELD OF THE INVENTION

This invention relates to an oscillating disc cutter with speed controlling bearings and has been devised particularly though not solely to prevent high speed rotation of a disc cutter when the cutting disc is disengaged from a rock face.

BACKGROUND OF THE INVENTION

Oscillating disc cutters of the type described in international patent specification ~~PCT/AU00/00066~~WO 00/46486 (the contents of which are incorporated herein by way of cross reference) have the general requirement that a mechanism is provided to prevent the cutting disc from rotating at a high speed when the cutter is not engaging the rock face. It should be noted that the reference to international patent specification ~~PCT/AU00/00066~~WO 00/46486 is not an admission that this publication forms part of the common general knowledge in Australia or in any other territory.

In normal cutting mode, when the disc cutter is presented to the cutting face the disc naturally rotates at about 30-40 rpm in the opposite direction to the shaft due to the rubbing friction caused by displacement difference between the diameter of the cutting disc and oscillating path diameter. It will be appreciated that this low speed rotation in the cutting mode is advantageous because it provides for even wear of the cutting disc and prevents temperature build-up at one point on the cutter.

However, during free running mode, when the cutter is not in contact with the rock face, torque transmitted to the disc from the shaft through bearing 609 (shown in FIG~~ig.ure~~ 7 of ~~PCT/AU00/00066~~WO 00/46486 and reproduced here as FIG. 5), causes the disc cutter to rotate in the same direction as the shaft. Without some degree of control, the cutter would speed up to around the same speed as the shaft, i.e. around 3000 rpm.

Reapplying the cutter to the rock face causes an almost instantaneous acceleration of the disc from around 3000 rpm in one direction to around 30-40 rpm in the opposite direction. This can cause significant wear and damage to the cutting edge.

In international patent specification ~~PCT/AU00/00066~~WO 00/46486, a solution is proposed of using a gear arrangement shown generally 616 in ~~Figure~~ FIG. ~~7~~5, (FIG. 7 of that specification).

Such a gear arrangement is heavy, prone to wear, maintenance issues, and causes additional drag when the cutter is engaged with the rock face.

It is an object of the present invention to overcome or ameliorate at least one of the disadvantages of the prior art, or to provide a useful alternative.

SUMMARY OF THE INVENTION

Accordingly the present invention provides an oscillating disc cutter including a cutting disc and a drive mechanism, the drive mechanism including a drive shaft to effect eccentric oscillation of the cutting disc and a radial bearing disposed to permit relative rotation between the drive shaft and the cutting disc, the cutter further including a first axial bearing disposed to react axial forces while accommodating induced rotation of the cutting disc when operatively engaged and to induce a rotational drag thereby limiting rotational speed of the cutting disc when free running.

Preferably, the cutter further includes a second bearing to induce a predetermined axial load in the first bearing.

Preferably, the second bearing substantially reacts the axial forces induced by the first bearing.

Preferably, the first bearing is a oil operated hydrostatic bearing and the second bearing is a fluid pressurised and lubricated bearing.

Preferably, pressure in the fluid bearing is maintained at a level such that a predetermined maximum running clearance in the hydrostatic bearing is maintained thereby inducing shear forces in the oil of the hydrostatic bearing. Preferably, the shear forces cause rotational drag in the bearing thereby limiting the rotational speed of the cutting disc in when free running.

Preferably, the fluid bearing ~~is~~ takes the form of a water-pressurised annulus.

Preferably, the limited rotational speed of the cutting disc is 0 to 100 rpm.

BRIEF DESCRIPTION OF THE DRAWINGS

Notwithstanding any other forms that may fall within its scope, one preferred form of the invention will now be described by way of example only with reference to the accompanying drawings in which:

F<u>IG.</u>~~ig~~ 1 is a cross sectional elevation through an oscillating disc cutter incorporating the present invention;

FIG.~~ig~~ 2 is cross sectional view of a variation of the disc cutter shown in FIG~~ig.ure~~ 1; ~~and~~

FIG.~~ig~~ 3A is a partial view of a hydrostatic bearing face in accordance with the invention;~~ and~~

FIG.~~ig~~ 3B is cross sectional view of the bearing face shown in FIG~~ig.ure~~ 3A;

FIG. 4A is a reproduction of FIG. 1 from WO 00/46486 and shows a part cross-sectional view of an oscillating disc cutting device taken;

FIG. 4B is a reproduction of FIG. 2 from WO 00/46486 and is an enlarged view of the cutting device of FIG. 1;

FIG. 4C is a reproduction of FIG. 3 from WO 00/46486 and is a schematic view of the action of the cutting device in excavating a rock face; and

FIG 5 is a reproduction of FIG. 7 from WO 00/46486 and shows a part cross-sectional view of an oscillating disc cutting device.

DETAILED DESCRIPTION OF THE INVENTION

FIG. ~~1~~ 4A is a cross-sectional view of an oscillating disc cutting device according to WO 00/46486~~the invention~~. The cutting device 10 includes a mounting assembly 11 and a rotary disc cutter 12. The mounting assembly 11 includes a mounting shaft 13 which is rotatably mounted within a housing 14, that can constitute or be connected to a large mass for impact absorption. The housing 14 thus, can be formed of heavy metal or can be connected to a heavy metallic mass. The shaft 13 is mounted within the housing 14 by a bearing 15~~, which can be of any suitable type and capacity. The bearing 15 is~~ mounted ~~in any suitable manner known to a person skilled in the art, such as~~ against a stepped section 16.

The housing 14 can have any suitable construction, and in one form includes a plurality of metal plates fixed together longitudinally of the shaft 13. Such an arrangement is shown in FIG. ~~2~~4B, and with this arrangement, applicant has found that a plurality of iron plates 17a and a single lead plate 17b provides effective impact absorption based on weight and cost considerations.

The shaft 13 is mounted for rotating motion about a central longitudinal axis AA. The shaft 13 includes a driven section 18 and a mounting section 19. The driven section 18 is connected to drive means 20 at the end thereof remote from the mounting section

by any suitable connectors, such as heavy duty threaded fasteners 21, while a seal 22 is applied between the facing surfaces of the mounting section and the drive means.

The drive means 20 can take any suitable form and the means shown in FIG. ~~1~~4A is a shaft that may be driven by a suitable engine or motor. The drive means 20 is mounted within the housing 14 by bearings 23, which are tapered roller bearings~~,~~ ~~although other types of bearings could also be employed~~. The bearings 23 are mounted against a stepped section 24 of the drive means 20 and against a mount insert 25 which is also stepped at 26. The mount insert 25 is fixed by threaded connectors 27 to the housing 14 and fixed to the mount insert 25 by further threaded connectors 28 is a sealing cap 29 which seals against the drive means 20 by seals 30. The sealing cap 29 also locates the outer race 31 of the bearings 23 by engagement therewith at 32, while a threaded ring 33 locates the inner race 34.

The mounting section 19 is provided for mounting of the disc cutter 12 and is ~~angularly~~ offset from the axis AA of the driven section 18~~, which generally will be approximately normal to the rock face being excavated~~. In this particular embodiment, the mounting section is also angularly offset from axis AA. The axis BB of the mounting section 19 is shown in FIG. 1 and it can be seen that the offset angle α~~.alpha.~~ is in the order of a few degrees only. The magnitude of the offset between axis AA and BB ~~angle .alpha.~~ determines the size of the oscillating ~~and nutating~~ movement of the disc cutter 12 whilst the magnitude of the angle α determines the degree of nutating movement~~and the angle .alpha. can be arranged as appropriate~~. In other embodiments, the axes AA and BB may be offset parallel such that the angle α is zero. Such a configuration provides only oscillation and no nutation.

The disc cutter 12 includes an outer cutting disc 35 that is mounted on a mounting head 36 by suitable connecting means, such as threaded connectors 37. The outer cutting disc 35 includes a plurality of tungsten carbide cutting bits 38 which are fitted to the cutting disc in any suitable manner. Alternatively, a tungsten carbide ring could be employed. The outer cutting disc can be removed from the cutting device for replacement or reconditioning, by removing the connectors 37.

The disc cutter 12 is rotatably mounted on the mounting section 19 of the mounting shaft 13. The disc cutter 12 is mounted by a tapered roller bearing 39, that is located by a step 40 and a wall 41 of the mounting head 36. An inclined surface 42 of the mounting head 36 is disposed closely adjacent a surface 43 of a mounting insert 44.

The surfaces 42 and 43 are spaced apart with minimum clearance to allow relative rotating movement therebetween and in this nutating embodiment, the surfaces have a spherical curvature, the centre of which is at the intersection of the axes AA and BB.

~~A seal 45 is located in a recess 46 of the surface 42 to seal against leakage of lubricating fluid from between the mounting shaft 13, and the housing 14 and the disc cutter 12. A channel 47 is also provided in the surface 42 outwardly of the seal 45 and ducts 48 connect the channel 47 to a further channel 49 and a further duct 50 extends from the channel 49 to the front surface 51 of the outer cutting disc 35. Pressurised fluid can be injected into the various channels and ducts through the port 52 and that fluid is used to flush the underside of the cutting disc 35 as well as the relative sliding surfaces 42 and 43.~~

The disc cutter 12 is rotatably mounted to the mounting section 19 of the mounting shaft 13 by the tapered roller bearing 39 and by a further tapered roller bearing 53. The bearing 53 is far smaller than the bearing 39 for the reason that the large bearing 39 is aligned directly in the load path of the disc cutter and thus is subject to the majority of the cutter load. The smaller bearing 53 is provided to pre-load the bearing 39.

~~The bearing 52 is mounted against the inner surface of the mounting shaft 13 and the outer surface of a bearing loading facility, comprising a nut 54 and a pre-loading shaft 55. Removal of the outer cutting disc 35 provides access to the nut 54 for adjusting the pre-load of the bearing 53.~~

~~The nutating movement of the disc cutter 12, occurs simultaneously with the oscillating motion and that nutating movement is movement in which a point on the cutting edge of the disc cutter is caused to move sinusoidally, in a cyclic or continuous manner as the disc cutter rotates.~~ The osillating~~is~~ movement of the disc cutter applies an impact load to the rock surface under attack, that causes tensile failure of the rock. With reference to FIG. ~~3~~4C, it can be seen that the motion of the disc cutter 12 brings the cutting tip or edge 58 into engagement under the oscillating movement at point 59 of the rock 56. Such oscillating movement results in travel of the disc cutter 12 in a direction substantially perpendicular to the axis AA. ~~The provision of simultaneous nutating movement causes the cutting edge 58 to strike the face 59 substantially in the direction S, so that a rock chip 60 is formed in the rock as shown.~~ Future chips are defined by the dotted lines 61. The action of the disc cutter 12 against the under face 59 is similar to

that of a chisel in developing tensile stresses in a brittle material, such as rock, which is caused effectively to fail in tension.

The direction S of impact of the disc cutter against the rock under face 59 is reacted through the bearing 39 and the direction of the reaction force is substantially along a line extending through the bearing 39 and the smaller bearing 53.

~~In a cutting device according to the invention, the~~The mass of the disc cutter is relatively much smaller than the mass provided for load absorption purposes. The load exerted on the disc cutter when it engages a rock surface under the oscillating/nutating movement, is reacted by the inertia of the large mass, rather than by the support structure.

~~The cutting device of the invention is preferably mounted for movement into the rock being excavated. Thus, the device can be mounted for example, on wheels or rails and it is preferred that the mounting facility be arranged to react the approximate average forces applied by the disc cutter, while the large absorption mass reacts the peak forces.~~

The oscillating disc cutter of the present invention is generally similar in configuration to that described above. More particularly, the disc cutter shown in ~~the accompanying drawing~~FIGS. 1 and 2 is generally similar in configuration to that shown in FIG~~ig.ure~~ 7 of international patent specification ~~PCT/AU00/00066~~WO 00/46486, and reproduced here as FIG. 5. ~~with like~~Like numbers refer~~ring~~ to the components in that drawing as described in the description of the international patent specification.

Instead of the bearings 605 and 606 ~~from PCT/AU00/00066~~ being water lubricated, only bearing 605 in the present invention is water lubricated. Bearing 606 is replaced by a hydrostatic bearing 700 supplied with high pressure oil through an annular passageway 701 inside a demountable ring 702, to which oil is supplied under pressure via nipple 703. The bearing 700 contains pockets 800 in the normal manner of hydrostatic bearings.

As can clearly be seen in ~~Figure~~ FIG 3A, these pockets may be in the form of a concentric grid pattern on the casing body opposing the disc 603, however, in alternative embodiments they may take on any form as is known in the art of hydrostatic bearings. In this embodiment there are ten pockets 800 evenly disposed in a circular array around the bearing. Each pocket's extremity is defined by a peripheral groove 801. A further oil channel groove in the form of a cross 802 dissects each pocket into four lands 803.

Referring to Figure 3B, these lands are at substantially the same height as the bearing surface between the pockets. Many hydrostatic bearings do not include these lands and the pockets are merely depressions in the bearing surface. However, in this embodiment, the lands effectively function to reduce the clearance gap between the bearing surfaces over a greater area thereby increasing the shear in the oil and enhancing the viscous drag characteristics of the bearing.

Oil is feed into the centre of each cross through a respective flow control orifice 706. Each respective orifice regulates the oil in each of the pockets of the bearing as is common in hydrostatic bearings.

Referring to ~~Figure~~ FIG. 2, oil exiting the bearing is able to seep either directly into the body of the device between bearings 609 and 610 or into outer drain channel 705 at the periphery of the bearing.

Providing a set minimum load on the hydrostatic bearing is fluid bearing 605. This fluid bearing maybe considered simply as a pressurised annulus, however, is referred to throughout as a fluid lubricated bearing. The fluid bearing surfaces include an annular plate portion of the disc 603 and a corresponding portion of the cutter housing opposing the annular plate. These bearing surfaces are separated by an annular gap into which water is introduced at pressure through a series of passageways 607. A hose and hose fittings (not shown) may be used to transport pressurised water from a pressure pump (not shown). In this embodiment the water is en-route to the cooling jets for the cutting edge of the cutter however, in other embodiments, separate cooling water and bearing water systems may be used. In still further embodiments, different fluids may be used for cooling and pressurising the fluid bearing.

The pressurised water provides a force on the plate thereby maintaining clearance between the bearing surfaces and providing an opposing force to the hydrostatic bearing. It will be appreciated that by regulating the pressure of the water, the magnitude of opposing force may also be regulated. Accordingly, by carefully controlling the water pressure in the fluid bearing and the oil pressure in the hydrostatic bearing, the clearance between the faces of the hydrostatic bearing can be set.

It will also be appreciated that the fluid bearing allows for a minute amount of axial yaw if the cutter head is differentially loaded. Such differential loading is accommodated and resisted by the hydrostatic bearing.

The fluid bearing surfaces may be covered with an antifriction material, as a safety measure should the bearing surfaces contact, for instance, as a result of failed water supply or during transport.

Typical values for the oil pressure supplied to the hydrostatic bearing and water pressure supplied to the fluid bearing are 14,000 kPa and 800 kPa respectively.

In operation, the cutter is powered by a 2-pole induction motor which, with a power supply at 50 Hz, rotates the dive shaft 612 at a speed of around 3000 rpm. Of course, alternative power supplies and a range of cutting speeds may be used.

However, it will be appreciated that drag inherent in the fluid and hydrostatic bearings provides a balancing torque to counter the rotation of the disc. By carefully selecting an appropriate pressure level in the fluid bearing, the clearance between the faces of the hydrostatic bearing are such that the rate of shear of the oil will rise with increasing speed of the disc. The friction developed due to the shear in the oil balances the rotation causing torque thereby limiting the free running speed of the disc to a desired value.

It will be appreciated that as well as rotation speed and clearance in the hydrostatic bearing, the frictional forces developed will also depend upon the design of the hydrostatic bearing surfaces and oil pockets and viscosity of the oil used. In turn, oil temperature will affect oil viscosity and therefore bearing performance. In this embodiment, standard hydraulic fluid is used however, other appropriate oils may be used as a replacement. The relationship between the viscosity of the oil selected and temperature is critical when selecting the oil.

Accordingly, the pressure of water supplied to the water lubricated bearing, the oil type, and the oil viscosity, temperature and pressure in the hydrostatic bearing are all carefully selected and controlled where appropriate to ensure correct function of the bearing and to avoid damage to the parts. In this regard the oil is passed through a heat exchanger of sufficient capacity to control the oil temperature.

An additional retardation force may be applied by drag inherent in the fluid bearing. Disengaging the cutter from the rock face reduces the axial load on the hydrostatic bearing which in turn causes the disc 603 to be forced closer to the water lubricated bearing surface 605. This may provide for an increase in drag thereby preventing the disc 603, to which the disc cutter 602 is bolted, from rotating at a high speed when the cutter is not engaging the rock face.

In this embodiment, the free running speed is selected to be about 30-40 rpm. While this is in the reverse direction to the operational speed, the difference is small enough to prevent damage and substantial wear to the cutter disc. However, in alternative embodiments, the parameters of the system may be selected to provide for virtually any free running speed desired in the direction of the shaft.

Accordingly, the drag in each axial bearing combines to eliminate the need for the gear arrangement 616 referred to in the description of FIG~~ig~~. 7 in international patent specification ~~PCT/AU00/00066~~WO 00/46486.

In alternative embodiments of the invention, other types of axial load bearings known in the art may replace the hydrostatic and fluid lubricated bearings. For instance, the hydrostatic bearing may be replaced by a Michell bearing and the fluid lubricated bearing may take to form of a mechanical, hydrodynamic, electromagnetic or other type of bearing able to withstand and/or provide an axial load. In such embodiments, one or other of the bearings may have a more significant effect in controlling the speed of the cutter disc when free spinning.

Although the cutting device is of the type generally described in WO 00/46486~~PCT/AU00/00030~~, it will be appreciated that various types of similar cutting devices may be used, with or without the nutating feature described in that patent specification.

It will be appreciated that the invention provides an effective means for limiting the speed of the cutter disc when in free running mode without the use of mechanical parts which are comparatively higher wearing.

Thus, in essence, the water lubricated bearing 605 and the hydrostatic bearing function as drag brakes on the rotation of the disc 603 and hence of the cutter 602.

Although the invention has been described with reference to specific examples it will be appreciated by those skilled in the art that the invention may be embodied in many other forms.